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DESCRIPTION

Variable Displacement Swash Plate Type Compressor

TECHNICAL FIELD

The present invention relates to a variable displacement swash plate type compressor that forms, for example, part of a refrigeration circuit and compresses refrigerant gas.

BACKGROUND ART

As shown in Fig. 9, such a swash plate type compressor includes a swash plate 92, which is coupled to a drive shaft 91 to be rotatable integrally with the drive shaft 91. Single head pistons 94 are coupled to the outer circumferential portion of the swash plate 92 with pairs of semispherical shoes 93A, 93B. Therefore, when the swash plate 92 is rotated by rotation of the drive shaft 91, the swash plate 92 slides with respect to the shoes 93A, 93B causing the pistons 94 to reciprocate, thereby compressing refrigerant gas.

Each pair of shoes 93A, 93B rotates about an axis S (a line that passes through the center of curvature P of the spherical surface and is perpendicular to sliding surfaces with respect to the swash plate 92) as the shoes 93A, 93B rotate relative to the swash plate 92. The rotation of the shoes 93A, 93B about the axis S is caused because a rotational force is applied to the shoes 93A, 93B in one direction about the axis S due to the difference between the circumferential velocities of the inner and outer circumferences of the swash plate 92. More specifically, the circumferential velocity of the outer circumference of the swash plate 92 is greater than that of the inner circumference of the swash plate 92.

That is, the swash plate type compressor shown in Fig. 9 is configured such that the shoes 93A, 93B directly slide against the swash plate 92. Therefore, the shoes 93A, 93B are unnecessarily rotated about the axis S due to the sliding motion caused as the shoes 93A, 93B rotate relative to the swash plate 92. This increases the mechanical loss particularly at the sliding portion between each piston 94 and the corresponding shoe 93B that receives reactive force of compression, and causes problems such as seizure at the sliding portions.

To solve such problems, for example, a technique shown in Fig. 10 has been proposed (for example, patent document 1). That is, an annular step 90a is provided at the center of a rear surface (a surface facing rightward in Fig. 10) of a swash plate (hereinafter, referred to as a first swash plate 90). An annular sliding plate (hereinafter, referred to as a second swash plate 95) is arranged outward of the step 90a of the first swash plate 90. The second swash plate 95 is supported to be coaxial with and rotatable relative to the first swash plate 90. The outer circumferential portion of the second swash plate 95 is arranged between the first swash plate 90 and the second shoes 93B to be slidable with respect to the first swash plate 90 and the second shoes 93B.

Therefore, when the first swash plate 90 is rotated, the first swash plate 90 slides relative to the second swash plate 95, which reduces the rotation speed of the second swash plate 95 as compared to the rotation speed of the first swash plate 90. This reduces the relative rotation speed of the second swash plate 95 and the second shoes 93B as compared to the relative rotation speed of the second shoes 93B and the first swash plate 90. As a result, the rotation of each second shoe 93B about the axis S caused by the

relative rotation of the second swash plate 95 and the second shoes 93B is suppressed, which suppresses mechanical loss and occurrence of problems.

A configuration has also been proposed in which rolling elements are provided between the first shoes 93A and the second shoes 93B and between the first swash plate 90 and the second swash plate 95 (for example, patent document 2). In the patent document 2, a race of a thrust bearing arranged toward the second shoe 93B can be considered as the second swash plate 95. With this configuration, the first swash plate 90 reliably slides with respect to the second swash plate 95, which significantly reduces the relative rotation speed of the second swash plate 95 and the second shoes 93B as compared to the relative rotation speed of the second shoes 93B and the first swash plate 90.

However, according to the swash plate configuration including the second swash plate 95 and the rolling element in addition to the first swash plate 90, the thickness between the first shoes 93A and the second shoes 93B is increased. The first swash plate 90, which tilts with respect to the drive shaft 91, has a salient corner 90b at the outer circumferential edge portion corresponding to the vicinity of the piston 94 located at the top dead center position (the state shown in Fig. 10). The salient corner 90b is provided at the outer circumferential edge portion opposite to the second swash plate 95 and significantly protrudes in the radial direction (upward in the drawing) of the drive shaft 91. Furthermore, the second swash plate 95, which tilts with respect to the drive shaft 91, has a salient corner 95b at the outer circumferential edge portion corresponding to the vicinity of the piston 94 located at the bottom dead center position (not shown). The salient corner 95b is provided at the outer circumferential edge portion opposite to the first

swash plate 90 and significantly protrudes in the radial direction of the drive shaft 91.

When the salient corner 90b of the first swash plate 90 and the salient corner 95b of the second swash plate 95 significantly protrude in the radial direction of the drive shaft 91, part of each piston 94 corresponding to the protruding portions needs to be made thin, or the pistons 94 need to be enlarged in the radial direction to avoid interference with the protruding portions. Reducing the thickness of the pistons 94 leads to reduction in the durability, and enlargement of the pistons 94 leads to enlargement of the swash plate type compressor. Therefore, in the prior art, when the thickness of the swash plate configuration needs to be increased, the radii of the first swash plate 90 and the second swash plate 95 are reduced to avoid interference of the salient corners 90b, 95b with the pistons 94.

However, when the radii of the first swash plate 90 and the second swash plate 95 are reduced, particularly the piston 94 located in the vicinity of the top dead center position (in a compression stroke) has a reduced contact area between the second shoe 93B, which receives a significant reaction force of compression, and the second swash plate 95. This undesirably reduces the durability of the second swash plate 95 and the second shoe 93B.

It has become a common practice to use carbon dioxide as refrigerant of the refrigeration circuit. When carbon dioxide refrigerant is used, the pressure in the refrigeration circuit becomes extremely high as compared to a case where chlorofluorocarbon refrigerant (for example, R134a) is used. Therefore, the reaction force of compression applied to the pistons 94 is increased in the swash plate type compressor,

and the aforementioned problem (reduction in the durability of the second swash plate 95 and the second shoes 93B) has become a significant matter of concern.

Patent Document 1: Japanese Laid-Open Patent Publication No. 8-338363 (page 4, Fig. 1)

Patent Document 2: Japanese Laid-Open Patent Publication No. 8-28447 (page 3, Fig. 1)

SUMMARY OF THE INVENTION

Accordingly, it is an objective of the present invention to provide a variable displacement swash plate type compressor that improves the durability of a swash plate and shoes while suppressing reduction in the durability of pistons and enlargement of the pistons.

To achieve the above objective, the present invention provides a variable displacement swash plate type compressor. A swash plate is coupled to a drive shaft to be rotatable integrally with the drive shaft. Pistons are coupled to the swash plate via shoes. Rotation of the drive shaft rotates the swash plate, which causes the pistons to reciprocate and compress gas. The displacement is changed by varying the inclination angle of the swash plate. An inclined surface is provided at part of the entire outer circumferential edge portion of the swash plate.

Providing the inclined surface at a projecting salient corner of the outer circumferential edge portion of the swash plate, which inclines with respect to the drive shaft, permits the diameter of the swash plate to be increased while suppressing decrease of the durability and enlargement of the pistons. Therefore, a significant reaction force of compression applied to the swash plate via the shoes is

received in a suitable manner. This improves the durability of the swash plate and the shoes.

In a preferred embodiment, part of the outer circumferential edge portion of the swash plate corresponding to the piston located at the top dead center position is provided with the inclined surface on a salient corner opposite to the piston. That is, part of the outer circumferential edge portion of the swash plate corresponding to a circumferential range of the swash plate that arranges any of the pistons at the top dead center position is provided with the inclined surface on the salient corner opposite to the piston.

At the outer circumferential edge portion of the swash plate that corresponds to the piston located at the top dead center position, the salient corner opposite to the piston significantly projects in the radial direction of the drive shaft when the swash plate tilts with respect to the drive shaft. Therefore, a significant reaction force of compression applied to the swash plate via the shoe of the piston located in the vicinity of the top dead center position is received in a suitable manner. This improves the durability of the swash plate and the shoes.

In a preferred embodiment, part of the outer circumferential edge portion of the swash plate corresponding to the piston located at the bottom dead center position is provided with the inclined surface on a salient corner toward the piston. That is, part of the outer circumferential edge portion of the swash plate corresponding to a circumferential range of the swash plate that arranges any of the pistons at the bottom dead center position is provided with the inclined surface on the salient corner toward the piston.

At the outer circumferential edge portion of the swash plate corresponding to the piston located at the bottom dead center position, the salient corner toward the piston significantly projects in the radial direction of the drive shaft. Therefore, chamfering the projecting portion of the swash plate permits the diameter of the first swash plate to be increased while suppressing decrease of the durability and enlargement of the pistons.

In the preferred embodiment, the swash plate includes a first swash plate, which is coupled to the drive shaft to be rotatable integrally with the drive shaft, and a second swash plate, which is supported by the first swash plate. The pistons are coupled to the first and second swash plates via first shoes, which abut against the first swash plate, and second shoes, which abut against the second swash plate and receive a reaction force of compression. Part of the outer circumferential edge of the first swash plate corresponding to the piston located at the top dead center position is provided with the inclined surface on a salient corner opposite to the second swash plate. That is, part of the outer circumferential edge portion of the first swash plate corresponding to a circumferential range of the first swash plate that arranges any of the pistons at the top dead center position is provided with the inclined surface on the salient corner opposite to the first swash plate.

At the outer circumferential edge portion of the first swash plate that corresponds to the piston located at the top dead center position, the salient corner opposite to the second swash plate significantly projects in the radial direction of the drive shaft when the first swash plate tilts with respect to the drive shaft. Therefore, chamfering the projecting portion of the first swash plate permits the diameter of the first swash plate to be increased while

suppressing decrease of the durability and enlargement of the pistons. Therefore, the first swash plate supports the second swash plate in a suitable manner, and a great reaction force of compression applied to the second swash plate via the second shoe of the piston located in the vicinity of the top dead center position is received by the first swash plate via the second swash plate in a suitable manner. This improves the durability of the second swash plate and the second shoes.

In the preferred embodiment, part of the outer circumferential edge portion of the first swash plate corresponding to the piston located at the bottom dead center position is provided with the inclined surface on a salient corner toward the second swash plate. That is, part of the outer circumferential edge portion of the first swash plate corresponding to a circumferential range of the first swash plate that arranges any of the pistons at the bottom dead center position is provided with the inclined surface on the salient corner toward the second swash plate.

At the outer circumferential edge portion of the swash plate corresponding to the piston located at the bottom dead center position, the salient corner toward the piston significantly projects in the radial direction of the drive shaft. Therefore, chamfering the projecting portion of the swash plate permits the diameter of the first swash plate to be increased while suppressing decrease of the durability and enlargement of the pistons.

In the preferred embodiment, the gas is refrigerant used in a refrigeration circuit, and carbon dioxide is used as the refrigerant.

When carbon dioxide refrigerant is used, as compared to a case where chlorofluorocarbon refrigerant (for example,

R134a) is used, the pressure in the refrigeration circuit becomes extremely high. Therefore, the reaction force of compression applied to the pistons in the variable displacement swash plate type compressor is increased, which increases the pressure between the swash plate and the shoes. The above mentioned embodiments of the present invention according to any one of claims 1 to 5 are particularly effective in improving the durability of the swash plate and the shoes while suppressing decrease of the durability and enlargement of the pistons.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a longitudinal cross-sectional view illustrating a variable displacement swash plate type compressor according to a first embodiment of the present invention;

Fig. 2 is an enlarged partial view of Fig. 1 with the first and second swash plates not being sectioned;

Fig. 3 is a longitudinal cross-sectional view illustrating a variable displacement swash plate type compressor according to a second embodiment of the present invention;

Fig. 4 is an enlarged partial view of Fig. 3 with the first and second swash plates not being sectioned (partially cut away) and part of the first and second shoes being sectioned;

Fig. 5 is an enlarged partial view illustrating a swash plate configuration according to a third embodiment of the present invention;

Fig. 6 is a longitudinal cross-sectional view illustrating a variable displacement swash plate type compressor according to a fourth embodiment of the present invention;

Fig. 7 is a cross-sectional view taken along line A-A of

Fig. 6;

Fig. 8 is an enlarged partial cross-sectional view of Fig. 6;

Fig. 9 is a longitudinal cross-sectional view illustrating a prior art variable displacement swash plate type compressor; and

Fig. 10 is a partial cross-sectional view illustrating a prior art technique.

BEST MODE FOR CARRYING OUT THE INVENTION

A variable displacement swash plate type compressor according to first to fourth embodiments of the present invention will now be described. The compressor forms part of a refrigeration circuit of a vehicle air-conditioning system.

The first embodiment will be described with reference to Figs. 1 and 2.

Fig. 1 is a longitudinal cross-sectional view of the variable displacement swash plate type compressor (hereinafter, simply referred to as the compressor) 10. The left end of the compressor 10 in Fig. 1 is defined as the front of the compressor 10, and the right end is defined as the rear of the compressor 10.

As shown in Fig. 1, a housing of the compressor 10 includes a cylinder block 11, a front housing member 12 secured to the front end of the cylinder block 11, and a rear housing member 14 secured to the rear end of the cylinder block 11 with a valve plate assembly 13 in between.

In the housing of the compressor 10, the cylinder block 11 and the front housing member 12 define a crank chamber 15. A drive shaft 16 is rotatably arranged between the cylinder

block 11 and the front housing member 12 and extends through the crank chamber 15. The drive shaft 16 is coupled to a power source of the vehicle, which is an engine E in this embodiment, through a clutchless type power transmission mechanism PT, which constantly transmits power. Therefore, the drive shaft 16 is always rotated by the power supply from the engine E when the engine E is running.

A rotor 17 is coupled to the drive shaft 16 and is located in the crank chamber 15. The rotor 17 rotates integrally with the drive shaft 16. The crank chamber 15 accommodates a substantially disk-like first swash plate 18. A through hole 18a is formed at the center of the first swash plate 18. The drive shaft 16 is inserted through the through hole 18a of the first swash plate 18. The first swash plate 18 is supported by the drive shaft 16 via the through hole 18a to be slidable and tiltable with respect to the drive shaft 16. A hinge mechanism 19 is located between the rotor 17 and the first swash plate 18.

The hinge mechanism 19 includes two rotor protrusions 41 (one of the protrusions 41 located toward the front of the sheet of Fig. 1 is not shown), which protrude from the rear surface of the rotor 17, and a swash plate protrusion 42, which protrudes from the front surface of the first swash plate 18 toward the rotor 17. The distal end of the swash plate protrusion 42 is inserted between the two rotor protrusions 41. Therefore, rotational force of the rotor 17 is transmitted to the first swash plate 18 via the rotor protrusions 41 and the swash plate protrusion 42.

A substantially cylindrical support portion 39 projects at the center of the rear surface of the first swash plate 18 to surround the drive shaft 16. A disk-like second swash plate 51 is arranged outward of the support portion 39 of the

first swash plate 18. A support hole 51a is formed at the center of the second swash plate 51. The support portion 39 is inserted in the support hole 51a. The radius of the second swash plate 51 is substantially the same as that of the first swash plate 18.

A radial bearing 52 is provided between the outer circumferential surface of the support portion 39 and the inner circumferential surface of the support hole 51a of the second swash plate 51. A thrust bearing 53 is provided between the rear surface of the first swash plate 18 and the front surface of the second swash plate 51. The thrust bearing 53 has rolling elements, which are rollers 53a in this embodiment, and the rollers 53a are rotatably held by a retainer 53b.

The second swash plate 51 is supported by the first swash plate 18 (the support portion 39) via the radial bearing 52 and the thrust bearing 53 such that the second swash plate 51 rotates relative to and tilt integrally with the first swash plate 18.

A cam portion 43 is formed at the proximal end of the rotor protrusions 41. A cam surface 43a is formed on the rear end face of the cam portion 43 facing the first swash plate 18. The distal end of the swash plate protrusion 42 slidably abuts against the cam surface 43a of the cam portion 43. Therefore, the hinge mechanism 19 guides the inclination of the first swash plate 18 and the second swash plate 51 as the distal end of the swash plate protrusion 42 moves toward and apart from the drive shaft 16 along the cam surface 43a of the cam portion 43.

Cylinder bores 22 are formed in the cylinder block 11 about the axis L of the drive shaft 16 at equal angular

intervals and extend in the front-rear direction (left-right direction on the sheet of Fig. 1). A single head piston 23 is accommodated in each cylinder bore 22 to be movable in the front-rear direction. The front and rear openings of each cylinder bore 22 are closed by the front end face of the valve plate assembly 13 and the associated piston 23. Each cylinder bore 22 defines a compression chamber 24. The volume of each compression chamber 24 changes according to the reciprocation of the corresponding piston 23.

Each piston 23 is formed by coupling, in the front-rear direction, a columnar head portion 37, which is inserted in the associated cylinder bore 22, and a neck 38 located in the crank chamber 15 outside the cylinder bore 22. The head portions 37 and the necks 38 are formed of an aluminum based metal material (pure aluminum or an aluminum alloy). A pair of shoe seats 38a are formed in each neck 38. Each neck 38 accommodates semispherical first and second shoes 25A, 25B. The first shoe 25A and the second shoe 25B are formed of iron based metal material. In this specification, "semisphere" refers not only to a half of a sphere, but also to a shape that includes part of a spherical surface of a sphere.

The first shoe 25A and the second shoe 25B are each received by the corresponding shoe seat 38a via a semispherical surface 25a. The semispherical surface 25a of the first shoe 25A and the semispherical surface 25a of the second shoe 25B are located on the same spherical surface defined about a point P. Each piston 23 is coupled to the outer circumferential portion of the first swash plate 18 and the second swash plate 51 via the first shoe 25A and the second shoe 25B. The first shoe 25A located opposite to the compression chamber 24 abuts against the front surface of the first swash plate 18 via a planar sliding surface 25b provided opposite to the semispherical surface 25a. The

second shoe 25B located toward the compression chamber 24, that is, the one that receives reaction force of compression abuts against the rear surface of the second swash plate 51 via a sliding surface 25b provided opposite to the semispherical surface 25a.

When the first swash plate 18 is rotated by the rotation of the drive shaft 16, the pistons 23 reciprocate in the front-rear direction.

When the first swash plate 18 is rotated, the radial bearing 52 and the thrust bearing 53 cause the first swash plate 18 to slide with respect to the second swash plate 51. This reduces the rotation speed of the second swash plate 51 as compared to the rotation speed of the first swash plate 18. Therefore, the relative rotation speed of the second swash plate 51 and the second shoes 25B is reduced as compared to the relative rotation speed of the second shoes 25B and the first swash plate 18. This suppresses the rotation of each second shoe 25B about the axis S (a line that passes through the center of curvature point P of the semispherical surface 25a and is perpendicular to the sliding surface 25b) caused by the relative rotation of the second swash plate 51 and the second shoe 25B. Thus, mechanical loss and occurrence of problems caused by the rotation of the second shoes 25B are suppressed.

An intake chamber 26 and a discharge chamber 27 are defined between the valve plate assembly 13 and the rear housing member 14 in the housing of the compressor 10. The valve plate assembly 13 includes intake ports 28 and intake valves 29 located between the compression chambers 24 and the intake chamber 26. The valve plate assembly 13 also includes discharge ports 30 and discharge valves 31 located between the compression chambers 24 and the discharge chamber 27.

As refrigerant of the refrigeration circuit, carbon dioxide is used. Refrigerant gas introduced into the intake chamber 26 from an external circuit, which is not shown, is drawn into each compression chamber 24 via the associated intake port 28 and the intake valve 29 as the corresponding piston 23 moves from the top dead center position to the bottom dead center position. The refrigerant gas that is drawn into the compression chamber 24 is compressed to a predetermined pressure as the piston 23 is moved from the bottom dead center position to the top dead center position, and is discharged to the discharge chamber 27 through the associated discharge port 30 and the discharge valve 31. The refrigerant gas in the discharge chamber 27 is then conducted to the external circuit.

A bleed passage 32, a supply passage 33, and a control valve 34 are provided in the housing of the compressor 10. The bleed passage 32 connects the crank chamber 15 to the intake chamber 26. The supply passage 33 connects the discharge chamber 27 to the crank chamber 15. The control valve 34, which is a conventional electromagnetic valve, is located in the supply passage 33.

The opening degree of the control valve 34 is adjusted by controlling power supply from the outside to control the balance between the flow rate of highly pressurized discharge gas supplied to the crank chamber 15 through the supply passage 33 and the flow rate of gas conducted out of the crank chamber 15 through the bleed passage 32. The pressure in the crank chamber 15 is thus determined. As the pressure in the crank chamber 15 varies, the difference between the pressure in the crank chamber 15 and the pressure in the compression chamber 24 is changed, which in turn varies the inclination angle of the first swash plate 18 and the second

swash plate 51. Accordingly, the stroke of each piston 23, or the compressor displacement is adjusted.

For example, when the opening degree of the control valve 34 is reduced, the pressure in the crank chamber 15 is reduced. Therefore, the inclination angle of the first swash plate 18 and the second swash plate 51 increases, thereby increasing the stroke of each piston 23. Thus, the displacement of the compressor 10 is increased. In contrast, when the opening degree of the control valve 34 increases, the pressure in the crank chamber 15 is increased. Therefore, the inclination angle of the first swash plate 18 and the second swash plate 51 is reduced, thereby reducing the stroke of each piston 23. Thus, the displacement of the compressor 10 is reduced.

As shown in Figs. 1 and 2, the support portion 39 of the first swash plate 18 supporting the second swash plate 51 is provided at a position decentered from the axis M1 of the first swash plate 18 toward the piston 23A located at the top dead center position. In other words, the support portion 39 is provided at a position decentered toward a section of the first swash plate (toward the hinge mechanism 19) that causes any of the pistons 23 to be located at the top dead center position as viewed in the radial direction of the first swash plate 18 from the axis M1. Therefore, the second swash plate 51, the radial bearing 52, and the thrust bearing 53 (and the retainer 53b) are decentered from the first swash plate 18 toward the piston 23A located at the top dead center position. Therefore, the axis M2 of the second swash plate 51, the radial bearing 52, and the thrust bearing 53 is slightly displaced in parallel from the axis M1 of the first swash plate 18 toward the center point P of the first shoe 25A and the second shoe 25B of the piston 23A located at the top dead center position (for example, 0.05 to 5 mm, although the

displacement is exaggerated in Figs 1 and 2).

Therefore, part of the outer circumferential edge portion of the second swash plate 51 corresponding to the vicinity of the piston 23A located at the top dead center position slightly protrudes in the radial direction of the first swash plate 18 from the outer circumferential edge portion of the first swash plate 18. Therefore, for example, as compared to a case where the second swash plate 51 is not decentered from the first swash plate 18, the contact area between the second shoe 25B of the piston 23 located in the vicinity of the top dead center position and the second swash plate 51 is increased.

Part of the outer circumferential edge portion of the second swash plate 51 corresponding to the vicinity of the piston 23B located at the bottom dead center position is located radially inward of the first swash plate 18 from the outer circumferential edge portion of the first swash plate 18. That is, part of the outer circumferential edge portion of the second swash plate 51 corresponding to the vicinity of the hinge mechanism 19 is located radially inward of the first swash plate 18 than the outer circumferential edge portion of the first swash plate 18. Therefore, for example, as compared to a case where the second swash plate 51 is not decentered from the first swash plate 18, the contact area between the second shoe 25B of the piston 23 located in the vicinity of the bottom dead center position and the second swash plate 51 is reduced. However, the reaction force of compression applied to the second shoe 25B of the piston 23 located in the vicinity of the bottom dead center position is far smaller than the reaction force of compression applied to the second shoe 25B of the piston 23 located in the vicinity of the top dead center position. Therefore, even if the contact area between the second shoe 25B of the piston 23

located in the vicinity of the bottom dead center position and the second swash plate 51 is reduced, no problem arises in the durability of the second swash plate 51 and the second shoe 25B.

Part of the outer circumferential edge portion of the first swash plate 18 corresponding to the piston 23A located at the top dead center position and circumferentially adjacent parts thereof are provided with an inclined surface (a chamfer) on a salient corner 18b opposite to the second swash plate 51. That is, part of the outer circumferential edge portion of the second swash plate 51 corresponding to the vicinity of the hinge mechanism 19 is provided with the inclined surface (the chamfer) on the salient corner 18b opposite to the second swash plate 51. In other words, part of the outer circumferential edge portion of the first swash plate 18 corresponding to a circumferential range of the first swash plate 18 that arranges any of the pistons 23 at the top dead center position is provided with the inclined surface on the salient corner 18b opposite to the piston 23A. The inclined surface (the chamfer) on the salient corner 18b is the largest at the part corresponding to the piston 23A located at the top dead center position, and gradually becomes smaller along the circumferential direction. The inclined surface (the chamfer) on the salient corner 18b is provided within a range of quarter to half the circumference of the first swash plate 18 with the part corresponding to the piston 23A located at the top dead center position arranged in the middle.

Part of the outer circumferential edge portion of the first swash plate 18 corresponding to the piston 23B located at the bottom dead center position and circumferentially adjacent parts thereof are provided with an inclined surface (a chamfer) on a salient corner 18c toward the second swash

plate 51. That is, part of the outer circumferential edge portion of the first swash plate 18 corresponding to a circumferential range of the first swash plate 18 that arranges the piston 23B at the bottom dead center position is provided with the inclined surface on the salient corner 18c opposite to the piston 23B.

The inclined surface (the chamfer) is the largest at the part corresponding to the piston 23B located at the bottom dead center position, and gradually becomes smaller along the circumferential direction. The inclined surface (the chamfer) of the salient corner 18c is provided within a range of quarter to half the circumference of the first swash plate 18 with the part corresponding to the piston 23B located at the bottom dead center position arranged in the middle. The inclined surface (the chamfer) on the salient corner 18c is substantially the same size as the inclined surface (the chamfer) on the salient corner 18b taking into consideration of the balance of the weight around the axis M1 of the first swash plate 18.

The first embodiment has the following advantages.

(1-1) The second swash plate 51 is decentered from the first swash plate 18 toward the piston 23A located at the top dead center position. Therefore, the contact area between the second shoe 25B of the piston 23 located in the vicinity of the top dead center position and the second swash plate 51 is increased without increasing the diameter of the first swash plate 18 and the second swash plate 51. Therefore, the second swash plate 51 reliably slides with respect to the second shoes 25B, and the durability of the second swash plate 51 and the second shoes 25B is improved while suppressing decrease of the durability and enlargement of the pistons 23.

(1-2) According to the swash plate configuration that includes the thrust bearing 53 in addition to the first swash plate 18 and the second swash plate 51 as in the first embodiment, the thickness of the swash plate configuration between the first shoes 25A and the second shoes 25B is increased. In such a configuration with a severe condition, decentering the second swash plate 51 with respect to the first swash plate 18 to increase the contact area between the second shoe 25B of the piston 23 located in the vicinity of the top dead center position and the second swash plate 51 is particularly effective in improving the durability of the second swash plate 51 and the second shoes 25B while suppressing decrease of the durability and the enlargement of the pistons 23.

(1-3) Part of the outer circumferential edge portion of the first swash plate 18 corresponding to the piston 23A located at the top dead center position is provided with the inclined surface on the salient corner 18b opposite to the second swash plate 51. Also, part of the outer circumferential edge portion of the first swash plate 18 corresponding to the piston 23B located at the bottom dead center position is provided with the inclined surface on the salient corner 18c toward the second swash plate 51. At the outer circumferential edge portion of the first swash plate 18 that corresponds to the piston 23A located at the top dead center position, the salient corner 18b opposite to the second swash plate 51 significantly projects in the radial direction of the drive shaft 16 when the first swash plate 18 tilts with respect to the drive shaft 16. Also, at the outer circumferential edge portion of the first swash plate 18 corresponding to the piston 23B located at the bottom dead center position, the salient corner 18c toward the second swash plate 51 significantly projects in the radial direction of the drive shaft 16.

Therefore, providing the inclined surfaces at the projecting portions of the first swash plate 18 (part of the entire circumference of the salient corners 18b, 18c) permits the diameter of the first swash plate 18 to be increased while suppressing decrease of the durability and enlargement of the pistons 23. Therefore, the first swash plate 18 supports the second swash plate 51 in a suitable manner, and a great reaction force of compression applied to the second swash plate 51 via the second shoe 25B of the piston 23 located in the vicinity of the top dead center position is received by the first swash plate 18 via the second swash plate 51 in a suitable manner. This improves the durability of the second swash plate 51.

(1-4) As the refrigerant of the refrigeration circuit, carbon dioxide is used. When carbon dioxide refrigerant is used, the pressure in the refrigeration circuit becomes extremely high as compared to a case where chlorofluorocarbon refrigerant (for example, R134a) is used. Therefore, the reaction force of compression applied to the pistons 23 in the compressor is increased, which increases the pressure between the second swash plate 51 and the second shoes 25B. The first embodiment of the present invention is thus particularly effective in improving the durability of the second swash plate 51 and the second shoes 25B while suppressing decrease of the durability and enlargement of the pistons 23.

Next, a second embodiment of the present invention will be described with reference to Figs. 3 and 4. In the second embodiment, only differences from the first embodiment are explained. Like or the same members are given the like or the same numbers and detailed explanations are omitted.

As for the first shoes 25A and the second shoes 25B, each first shoe 25A located toward the hinge mechanism 19, or opposite to the associated compression chamber 24, slidably abuts against the front surface of an outer circumferential portion18-1 of the first swash plate 18 via the sliding surface 25b opposite to the semispherical surface 25a. Also, each second shoe 25B located opposite to the hinge mechanism 19, or toward the associated compression chamber 24, and receives the reaction force of compression slidably abuts against the rear surface of an outer circumferential portion 51-2 of the second swash plate 51 via the sliding surface 25b opposite to the semispherical surface 25a. The center portion of the sliding surface 25b of the first shoe 25A bulges toward the first swash plate 18 (see Fig. 4. The bulge is exaggerated in Fig. 4). The sliding surface 25b of the second shoe 25B is flat.

A radial bearing 52A, which is a roller bearing, is located between the support portion 39, which forms the inner circumferential portion of the first swash plate 18, and an inner circumferential portion 51-1 of the second swash plate 51, and more specifically, between the outer circumferential surface of the support portion 39 and the inner circumferential surface of the support hole 51a of the second swash plate 51. The radial bearing 52A includes an outer race 52a attached to the inner circumferential surface of the support hole 51a of the second swash plate 51, an inner race 52b attached to the outer circumferential surface of the support portion 39 of the first swash plate 18, and rolling elements, which are rollers 52c in the second embodiment. The rollers 52c are located between the outer race 52a and the inner race 52b.

The thrust bearing 53, which is a roller bearing, is located between the first shoes 25A and the second shoes 25B

and between the outer circumferential portion 18-1 of the first swash plate 18 and the outer circumferential portion 51-2 of the second swash plate 51. The thrust bearing 53 has rolling elements, which are the rollers 53a in the second embodiment, and the rollers 53a are rotatably held by the retainer 53b. The thrust bearing 53 has an annular race 55 located between the rollers 53a and the first swash plate 18. The race 55 is formed by carburizing and heat treating base material formed of mild steel such as SPC. The corners at both ends of each roller 53a are chamfered to prevent the second swash plate 51 and the race 55 from being damaged by the rollers 53a abutting against the second swash plate 51 and the race 55.

An annular engaging portion 18d is provided on the rear surface of the first swash plate 18 at the outermost circumference of the outer circumferential portion 18-1 and projects toward the second swash plate 51. The race 55 is located inward of the engaging portion 18d and is engaged with the first swash plate 18 at the radially outward edge of the race 55 by the abutment between the outer circumferential edge of the race 55 and the engaging portion 18d. The race 55 is guided by the engaging portion 18d to rotate relative to the first swash plate 18.

The second swash plate 51 is supported by the first swash plate 18 via the radial bearing 52A and the thrust bearing 53 such that the second swash plate 51 rotates relative to and tilts integrally with the first swash plate 18. Therefore, when the first swash plate 18 is rotated, the radial bearing 52A and the thrust bearing 53 cause rolling motion between the first swash plate 18 and the second swash plate 51. Therefore, the mechanical loss caused by sliding motion between the first swash plate 18 and the second swash plate 51 is converted to the mechanical loss caused by the

rolling motion. This significantly suppresses the mechanical loss in the compressor.

The plate thickness Y1 of the inner circumferential portion 51-1 of the second swash plate 51 that is supported by the radial bearing 52A is greater than the plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 that is supported by the thrust bearing 53. More specifically, the plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 is half or more of the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18 and thinner than the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18. Also, the plate thickness Y1 of the inner circumferential portion 51-1 of the second swash plate 51 is thicker than the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18.

The plate thickness of the inner circumferential portion 51-1 of the second swash plate 51 is designed to be greater than that of the outer circumferential portion 51-2 of the second swash plate 51 ($Y1 > Y2$) by providing a cylindrical first projection 56, which projects toward the first swash plate 18, and a cylindrical second projection 57, which projects opposite to the first swash plate 18. The first projection 56 and the second projection 57 are arranged coaxial with the support hole 51a, and the inner circumferential surfaces of the first projection 56 and the second projection 57 form part of the inner circumferential surface of the support hole 51a. The outer diameter Z2 of the second projection 57 is smaller than the outer diameter Z1 of the first projection 56. Also, the outer circumferential corner 57a of the distal end face of the second projection 57 is entirely chamfered to form a tapered face.

The second embodiment provides the following advantages in addition to the advantages of the first embodiment.

(2-1) The thrust bearing 53, which supports the second swash plate 51 to be rotatable relative to the first swash plate 18, is arranged between the first shoes 25A and the second shoes 25B and between the outer circumferential portion 18-1 of the first swash plate 18 and the outer circumferential portion 51-2 of the second swash plate 51. The radial bearing 52A, which supports the second swash plate 51 to be rotatable relative to the first swash plate 18, is arranged between the inner circumferential portion (the support portion 39) of the first swash plate 18 and the inner circumferential portion 51-1 of the second swash plate 51.

Therefore, the thrust bearing 53 and the radial bearing 52A effectively reduce the rotational resistance caused between the outer circumferential portion 18-1 of the first swash plate 18 and the outer circumferential portion 51-2 of the second swash plate 51, and between the inner circumferential portion (the support portion 39) of the first swash plate 18 and the inner circumferential portion 51-1 of the second swash plate 51. Therefore, even in the compressor 10 used for the refrigeration circuit that uses carbon dioxide as refrigerant, the sliding motion between the first swash plate 18 and the second swash plate 51 is converted to the mechanical loss caused by the rolling motion. As a result, problems such as the mechanical loss and the seizure are effectively suppressed.

(2-2) The plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 is half or more of the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18 and

thinner than the plate thickness X of the outer circumferential portion 18-1. To avoid enlargement of the pistons 23, that is, enlargement of the compressor, a space between the first shoes 25A and the second shoes 25B is limited. In this limited space, when the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18 is increased, the plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 needs to be reduced. In contrast, when the plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 is increased, the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18 needs to be reduced.

In terms of receiving the reaction force of compression, the plate thicknesses X, Y2 of the outer circumferential portions 18-1, 51-2 of the first swash plate 18 and the second swash plate 51 need to be as thick as possible to secure the strength. However, securing the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18 to which power is transmitted from the drive shaft 16 should take precedence to securing the plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 that is only required to slide with respect to the first swash plate 18. In this respect, it is suitable to set the plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 to be half or more of the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18 and thinner than the plate thickness X of the outer circumferential portion 18-1.

(2-3) In the second swash plate 51, the plate thickness Y1 of the inner circumferential portion 51-1 is greater than the plate thickness Y2 of the outer circumferential portion 51-2. The thick inner circumferential portion 51-1 permits

the second swash plate 51 to be stably supported by the radial bearing 52A, and improves the sliding performance between the first swash plate 18 and the second swash plate 51. Furthermore, since the outer circumferential portion 51-2 of the second swash plate 51 is relatively thinner than the inner circumferential portion 51-1, the plate thickness of the outer circumferential portion 18-1 of the first swash plate 18 that is required to have a greater strength than the second swash plate 51 is easily secured.

(2-4) The plate thickness Y2 of the outer circumferential portion 51-2 of the second swash plate 51 is thinner than the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18. Therefore, the thin outer circumferential portion 51-2 of the second swash plate 51 facilitates securing the plate thickness of the outer circumferential portion 18-1 of the first swash plate 18 that is required to have a greater strength than the second swash plate 51. The plate thickness Y1 of the inner circumferential portion 51-1 of the second swash plate 51 is greater than the plate thickness X of the outer circumferential portion 18-1 of the first swash plate 18. Therefore, the radial bearing 52A more stably supports the second swash plate 51.

(2-5) As for the first projection 56 and the second projection 57, which form the inner circumferential portion 51-1 of the second swash plate 51, the outer diameter Z2 of the second projection 57 is less than the outer diameter Z1 of the first projection 56. When the displacement of the compressor 10 is maximum (state shown in Fig. 3), for example, part of the second projection 57 significantly approaches the piston 23B located at the bottom dead center position. Therefore, it is effective to make the diameter of the second projection 57 to be smaller than that of the first projection

56, thereby separating the second projection 57 from the piston 23, in view of avoiding interference between the second swash plate 51 and the pistons 23 while increasing the plate thickness Y1 of the inner circumferential portion 51-1 of the second swash plate 51.

(2-6) As for the second projection 57, which forms the inner circumferential portion 51-1 of the second swash plate 51, the outer circumferential corner 57a of the distal end face is chamfered. When the displacement of the compressor is maximum, for example, part of the outer circumferential corner 57a of the distal end face of the second projection 57 significantly approaches the piston 23B located at the bottom dead center position. Therefore, it is effective to provide the chamfer on the outer circumferential corner 57a of the distal end face of the second projection 57 in view of avoiding interference between the second swash plate 51 and the pistons 23 while increasing the plate thickness Y1 of the inner circumferential portion 51-1 of the second swash plate 51.

(2-7) Part of the outer circumferential edge of the first swash plate 18 corresponding to the piston 23A located at the top dead center position is provided with the inclined surface (the chamfer) on the salient corner 18b opposite to the second swash plate 51. Therefore, the first swash plate 18 and the second swash plate 51 can be enlarged while suppressing reduction in the durability and enlargement of the pistons 23. Therefore, the second swash plate 51 reliably slides with respect to the second shoes 25B, and the durability of the second swash plate 51 and the second shoes 25B is improved while suppressing reduction in the durability and enlargement of the pistons 23.

That is, at the outer circumferential edge portion of

the first swash plate 18 that corresponds to the piston 23A located at the top dead center position, the salient corner 18b (that has not been chamfered) opposite to the second swash plate 51 significantly projects in the radial direction of the drive shaft 16 when the first swash plate 18 tilts with respect to the drive shaft 16. When the salient corner 18b of the first swash plate 18 opposite to the second swash plate 51 significantly projects in the radial direction, the thickness of the necks 38 of the pistons 23 need to be reduced corresponding to the projecting portion, or the necks 38 need to be enlarged in the radial direction to avoid interference with the projecting portion. However, reducing the thickness of the necks 38 leads to reduction in the durability of the pistons 23, and enlargement of the necks 38 leads to enlargement of the compressor.

To solve such problems, the radius of the first swash plate 18 may be reduced to avoid interference between the salient corner 18b and the pistons 23. However, when the radius of the first swash plate 18 is reduced, the radius of the second swash plate 51, which needs to be supported by the first swash plate 18, must also be reduced. Therefore, in particular, the contact area between the second swash plate 51 and the second shoe 25B of the piston 23 located in the vicinity of the top dead center position (in the compression stroke) that receives a significant reaction force of compression is reduced, which reduces the durability of the second swash plate 51 and the second shoes 25B.

(2-8) As the rolling elements of the radial bearing 52A, the rollers 52c are used. The roller bearing that uses the rollers 52c as the rolling elements has superior load bearing properties as compared to, for example, a case where balls are used as the rolling elements. This reduces the size of the radial bearing 52A, which reduces the size of the

compressor 10.

(2-9) The race 55 is located between the rollers 53a of the thrust bearing 53 and the first swash plate 18. The race 55 is rotatable relative to the first swash plate 18.

In a case of a configuration in which, for example, the rollers 53a of the thrust bearing 53 roll directly on the first swash plate 18, a significant reaction force of compression is concentrated on part of the first swash plate 18 (part of the first swash plate 18 corresponding to the piston 23 located in the vicinity of the top dead center position), which may cause partial wear and deterioration. However, in the second embodiment, since the race 55 is provided between the rollers 53a and the first swash plate 18, the reaction force of compression applied to the rollers 53a is applied to the first swash plate 18 with reduced contact pressure via the race 55. Therefore, the first swash plate 18 is suppressed from being partially worn and deteriorated. Also, as for the race 55 that rotates relative to the first swash plate 18, the section to which a significant reaction force of compression is applied via the rollers 53a is sequentially changed. This prevents the race 55 from being partially worn and deteriorated.

(2-10) The engaging portion 18d is provided on the outer circumferential portion 18-1 of the first swash plate 18 and extends toward the second swash plate 51. The race 55 is engaged with the first swash plate 18 by abutting against the engaging portion 18d at the radially outward edge of the race 55.

For example, in a configuration in which the engaging portion is provided at the inner circumferential portion of the first swash plate 18 and the race 55 is engaged with the

first swash plate 18 at the radially inward edge, when lubricant (refrigerant oil) that is adhered to the first swash plate 18 moves radially outward by centrifugal force, the engaging portion hinders the lubricant from entering between the first swash plate 18 and the race 55. However, the second embodiment in which the race 55 is engaged with the first swash plate 18 at the radially outward edge prevents the engaging portion 18d from hindering the lubricant from entering between the first swash plate 18 and the race 55. Thus, the first swash plate 18 reliably slides with respect to the race 55.

(2-11) The engaging portion 18d has an annular shape. Therefore, the engaging portion 18d is stably engaged with the race 55. Thus, the race 55 further reliably slides with respect to the first swash plate 18.

Next, a third embodiment of the present invention will be described with reference to Fig. 5. In the third embodiment, only differences from the second embodiment are explained. Like or the same members are given the like or the same numbers and detailed explanations are omitted.

In the third embodiment, the support portion 39 is not decentered from the axis M1 of the first swash plate 18. That is, the second swash plate 51, the radial bearing 52A (see Fig. 3), and the thrust bearing 53 (including the race 55) are not decentered from the first swash plate 18. In this case, as for part of the outer circumferential edge of the first swash plate 18 that corresponds to the piston 23B located at the bottom dead center position, the salient corner 18c need not be chamfered as shown in Fig. 5 because the salient corner 18c toward the second swash plate 51 does not significantly project in the radial direction from the second swash plate 51.

Furthermore, in the third embodiment, the PCD of the thrust bearing 53 is greater than the diameter of an imaginary cylinder defined about the axes M1, M2 of the first swash plate 18 and the second swash plate 51 and passes through the center points P of the first shoe 25A and the second shoe 25B. In this manner, the thrust bearing 53 (the rollers 53a) receives the reaction force of compression transmitted through the second swash plate 51 in a suitable manner, which improves the durability. The "PCD" of the thrust bearing 53 refers to the diameter of an imaginary cylinder having the axis at the center of the thrust bearing 53 (at the axes M1, M2 of the first swash plate 18 and the second swash plate 51) and passes through the mid point of the rotating axis of the rollers 53a.

Next, a fourth embodiment of the present invention will be described with reference to Figs. 6 to 8. In the fourth embodiment, only differences from the first and second embodiments are explained. Like or the same members are given the like or the same numbers and detailed explanations are omitted.

The rotor 17 is fixed to the drive shaft 16, and a swash plate 58 is supported on the drive shaft 16. The swash plate 58 is permitted to slide along and incline with respect to the drive shaft. Coupling pieces 59, 60 are fixed to the swash plate 58, and guide pins 61, 62 are fixed to the coupling pieces 59, 60. A pair of guide holes 171 (only one is shown) is formed in the rotor 17. Head portions of the guide pins 61, 62 are slidably fitted to the guide holes 171. The engagement of the guide holes 171 with the guide pins 61, 62 allows the swash plate 58 to incline with respect to the axial direction of the drive shaft 16 and rotate integrally with the drive shaft 16. The inclination of the swash plate

58 is guided by the guide holes 171 and the guide pins 61, 62, and the drive shaft 16. The coupling pieces 59, 60, the guide pins 61, 62, and the guide holes 171 form a hinge mechanism 19A.

The swash plate 58 shown by a solid line in Fig. 6 is in the maximum inclination state of the swash plate 58. When the center of the swash plate 58 moves toward the cylinder block 11, the inclination of the swash plate 58 decreases. The swash plate 58 shown by a chain line in Fig. 6 is in the minimum inclination state.

Part of the outer circumferential edge portion of the swash plate 58 corresponding to the piston 23A located at the top dead center position and circumferentially adjacent parts thereof are provided with an inclined surface on a salient corner 58a opposite to the piston 23. That is, part of the outer circumferential edge portion of the swash plate 58 corresponding to the vicinity of the hinge mechanism 19A is provided with the inclined surface on the salient corner 58a toward the hinge mechanism 19A. In other words, part of the outer circumferential edge portion of the swash plate 58 corresponding to a circumferential range of the swash plate 58 that arranges the piston 23A at the top dead center position is provided with the inclined surface on the salient corner 58a opposite to the piston 23. As shown in Fig. 7, part of the inclined surface of the salient corner 58a corresponding to the piston 23 located at the top dead center position is the largest, and gradually becomes smaller along the circumferential direction.

As shown in Fig. 8, when the swash plate 58 is in the maximum inclination state, the inclined surface provided on the salient corner 58a is located on the circumferential surface of an imaginary cylinder C having an axis M3 that is

parallel to the axis L of the drive shaft 16. In the example shown in Fig. 8, the axis M3 is displaced with respect to the axis L from the piston 23A located at the top dead center position toward the drive shaft 16. The diameter of the imaginary cylinder C is greater than or equal to the diameter of the swash plate 58.

At the outer circumferential edge portion of the swash plate 58 that corresponds to the piston 23A located at the top dead center position, the salient corner 58a opposite to the piston 23 significantly projects in the radial direction of the drive shaft 16 when the swash plate 58 tilts with respect to the drive shaft 16. Therefore, providing the inclined surface at the projecting portion (part of the salient corner 58a) of the swash plate 58 permits the swash plate 58 to be enlarged while suppressing reduction in the durability and enlargement of the pistons 23. Therefore, a significant reaction force of compression applied to the swash plate 58 is received in a suitable manner via the second shoe 25B of the piston 23 located in the vicinity of the top dead center position. This improves the durability of the swash plate 58.

It should be understood that the invention may be embodied in the following forms without departing from the spirit or scope of the invention.

(1) In the first embodiment, the radial bearing 52 may be omitted, and the second swash plate 51 may slide with respect to the support portion 39.

(2) In the first embodiment, the thrust bearing 53 may be omitted, and the second swash plate 51 may directly slide with respect to the first swash plate 18.

(3) In the first embodiment, the radial bearing 52 and the thrust bearing 53 may be omitted, and the second swash plate 51 may be secured to the first swash plate 18 so that the second swash plate 51 rotates integrally with the first swash plate 18.

In this case, part of the outer circumferential edge portion of the second swash plate 51 corresponding to the piston 23A located at the top dead center position is provided with an inclined surface (a chamfer) on the salient corner toward the first swash plate 18. In addition, part of the outer circumferential edge portion of the second swash plate 51 corresponding to the piston 23B located at the bottom dead center position is provided with an inclined surface (a chamfer) on the salient corner opposite to the first swash plate 18.

With reference to Fig. 2, when the second swash plate 51 inclines with respect to the drive shaft 16, the salient corner toward the first swash plate 18 significantly projects in the radial direction of the drive shaft 16 at the outer circumferential edge portion of the second swash plate 51 that corresponds to the piston 23A located at the top dead center position. Also, at the outer circumferential edge portion of the second swash plate 51 corresponding to the piston 23B located at the bottom dead center position, the salient corner opposite to the first swash plate 18 significantly projects in the radial direction of the drive shaft 16. Therefore, providing the inclined surfaces (the chamfers) at the projecting portions (part of the salient corners) of the second swash plate 51 permits the second swash plate 51 to be enlarged while suppressing reduction in the durability and enlargement of the pistons 23. Therefore, the contact area between the second shoe 25B of the piston 23 located in the vicinity of the top dead center position and

the second swash plate 51 can further be increased, which further improves the durability of the second swash plate 51 and the second shoe 25B.

(4) In the first embodiment, two swash plates, which are the first swash plate 18 and the second swash plate 51, are used. However, for example, a third swash plate may be arranged between the second swash plate 51 and the second shoes 25B. That is, the swash plate configuration to which the present invention may be applied is not limited to the one that uses the first swash plate and the second swash plate, but the swash plate configuration may include a number of swash plates such as three, four, or five swash plates.

(5) The present invention may be applied to a variable displacement swash plate type compressor including double head pistons. In this case, the second swash plate may be arranged on either the front or rear surfaces of the first swash plate, or may be arranged on each of the front and rear surfaces of the first swash plate.

(6) The present invention need not be applied to the refrigerant compressor of the refrigeration circuit, but may be applied to, for example, an air-compressor.

(7) The second embodiment may be modified such that, for example, the sliding surface 25b of each first shoe 25A is flat as shown in Fig. 5.

(8) The second embodiment may be modified such that, for example, the sliding surface 25b of each second shoe 25B is dented at the center as shown in Fig. 5. In this case, the weight of each second shoe 25B, which reciprocate with the associated piston 23, is reduced, which reduces the inertial force of the second shoe 25B. Therefore, the inclination

angle of the first swash plate 18 and the second swash plate 51, that is, the displacement of the compressor is smoothly changed.

(9) In the second and third embodiments, the thrust bearing 53 may be changed to a roller bearing, which includes balls as the rolling elements.

(10) In the second and third embodiments, the thrust bearing 53 may be changed to a sliding bearing.

(11) In the second and third embodiments, the radial bearing 52A only receives a radial load (a load perpendicular to the axis M2) applied to the second swash plate 51. Instead, for example, the rollers 52c may be tilted with respect to the axis M2 of the second swash plate 51 such that the radial bearing 52A also receives a thrust load (a load along the axis M2) in addition to the radial load.

(12) In the second and third embodiments, the thrust bearing 53 only receives the thrust load applied to the second swash plate 51. Instead, for example, the rollers 53a may be tilted with respect to the surface of the second swash plate 51 such that the thrust bearing 53 also receives the radial load in addition to the thrust load.

(13) In the second and third embodiments, the race 55 may be omitted, and the rollers 53a of the thrust bearing 53 may roll directly on the first swash plate 18.

(14) In the second and third embodiments, the engaging portion 18d may be omitted, and an engaging portion may be provided on the inner circumferential portion of the first swash plate 18 (for example, the proximal portion of the support portion 39 may serve also as the engaging portion) so

that the race 55 is engaged with the first swash plate 18 on at radially inward edge.